Avoiding Wind Vibration Damage to Aerials

by Arthur Bolton, GM3BMI

INTRODUCTION

AN EXCELLENT PAPER (1) has already set out the way to design aerials and masts for static strength against wind forces, as well as giving much practical advice. If the guidance offered therein is followed, aerial damage by static wind forces should be avoided. Unfortunately aerials may be set into vibration by the wind and the alternating stresses for vibration are increased by a dynamic magnifier, which has exactly the same mathematical basis as the Q value of a tuned circuit, and may have a value of 60 or 70 times. This article is intended to extend G3ZPF's paper by describing the dynamic behaviour of typical aerials under the action of wind forces and to show how damage arising from vibration might be avoided.

The usual and most serious dynamic problem arises from vortex shedding, which can set an aerial into prolonged vibration across the wind. This can cause rapid failure when the wind speed rises during a big storm and the amplitude exceeds the failure limit. Failure in the first big storm shows that the strength was insufficient for the combined static (D.C.) and dynamic (A.C.) loads in spite of the fact that the calculations concerned with static conditions include an allowance for gusts. The trouble is that the allowance is only 10% or so when the dynamic magnifier for vibrations in 6000%.

Vortex shedding might alternatively give rise to failure after a relatively short time in service, during which wind speeds were never more than moderate. In this case metal fatigue is the culprit and arises from repeated loading at stresses far less than would be required to fail the element or mast if applied only once.

Fatigue cracks often start at surface scratches, holes, clamps or changes or cross-section, and then progress across the whole member until, for example, half an element falls off. The failure surface often shows 'tidal marks' where the failure progressed through the metal by fits and starts. It is known that fatigue arises from hundreds of thousands of repetitions of small stresses. Notice that if an aerial element vibrates in the wind at 10Hz this is 600 repetitions per minute or well over 400,000 repetitions when the wind has blown at the `awkward' speed for as little as twelve hours in total. This awkward wind speed is the one which sets the aerial into oscillation and might be only five or six metres/second rather than the full gale force which could occur at the site.

The ideal solution to the problem of vortex shedding is at the design stage by avoiding any resonances which might be excited by the wind. This will not always be possible and any existing aerial which presently vibrates in the wind also needs to be protected. The remedies available are of three kinds: to reduce the amount of excitation fed in by the wind, if possible to zero; to damp out whatever energy is unavoidably fed in by using suitable devices; or to move the frequency of any resonance to a less troublesome value by changing the diameter or wall thickness of the member concerned, or adding a brace to prevent that mode.

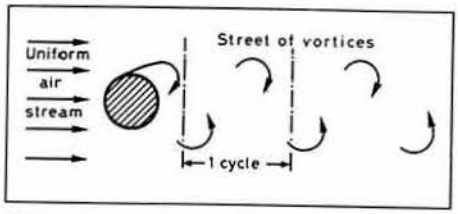


Fig 1.

VORTEX SHEDDING

The wind and the aerial can form a self-excited oscillator. Vortices are shed from the elements or the mast as in **Fig 1**, and trail downwind imparting a series of synchronous force pulses to an element or the mast as each vortex breaks away. These pulses can maintain the vibrations in the member if the shedding frequency is within about 20% of the natural frequency of vibration of the element. The frequency of vortex shedding is:-

$$f_v = v \times S / D \qquad (1)$$

where:-

f_v is the frequency (Hz) of shedding pairs of vortices, one from each 'edge' of the section,

v is the wind velocity (metres per second), perpendicular to a horizontal element, or in any direction for a vertical mast,

S is the non-dimensional Strouhal number, 0.2 for cylindrical sections and about 0.13 for most 'structural' sections, and

D is the depth (metres) of the section facing the wind (the diameter for tubular members).

The use of a consistent system of units has been emphasised to prevent errors in calculations concerned with mechanical vibrations, eg the metre for length, kilogram for mass, Newton for force and Hz for frequency. It might seem natural to use centimetres or millimetres especially for areas or the physical constants of materials but this often leads to errors and is best avoided. It might seem even more natural to use pounds and feet or inches but the system listed really is better for vibration and easier after a little practice.

It will be seen from equation (1) that as the wind speed rises the frequency of vortex shedding increases proportionately. The owner of an aerial needs to consider a frequency band stretching from zero to the frequency generated at the highest wind speed possible at the site, because the wind might blow at any speed in between. For example, if an aerial mast is a pole of 50mm external diameter and the maximum wind speed is 40 metres/second the band of vortex frequencies stretches from zero to:-

$$f_v = 40 \text{ m/s} \times 0.20 / 0.050\text{m} = 160\text{Hz}$$

This shows that the band of exciting frequencies is very wide and could lead to vibration as the wind speed rises through each successive natural frequency of the mast. Over such a range it might possibly excite any of the lowest three or four natural frequencies. This indicates the difficulty of dealing with vortex shedding by bracing or other structural design techniques alone. It points up the need to consider spoilers of some kind to prevent vortex shedding as far as possible, and dampers to reduce any remaining amplitudes of vibration.

The size of the dynamic forces exerted on the aerial.

The amplitude of the vortex shedding force depends on the square of the wind velocity, the geometry of the section and the density of air. With a safety allowance for different sectional shapes, and converting the force to Newtons per metre run of element for convenience in the later calculations, it works out as about:-

$$F = 0.33 \, D \times v^2 \text{ N/m} \qquad (2)$$

Suppose the 50mm mast in the previous example were 5m high, the maximum amplitude of force acting on the mast would be:-

$$F = 0.33 \times 0.050 \times 40^2 = 26.7 \text{ N/m}$$

This is about 134 N overall (over 30lbs in Imperial units) repeatedly applied from side to side and, if the mast had a natural frequency close to 160Hz, would surely destroy it, even though it met the static requirements of reference [1].

Coupling of the vortex shedding forces to the mechanical structure of the element, boom or mast.

The vortex shedding forces are only able to exert their damaging effect if they are in tune with a mechanical vibration of the aerial. If the natural resonant frequency of vibration of the aerial is close to the vortex shedding frequency being generated by the wind, the forces produced by vortex shedding will maintain vibration of the aerial at that natural frequency. This is much like the way in which a tuned circuit can couple to a tuned circuit in the next stage as long as both are tuned to the right frequency. If one circuit is mistuned the amplitude will be reduced or may even disappear.

The mechanical vibration depends upon the closeness of the natural frequency to the forcing frequency of the vortex shedding. If the two frequencies are identical, experiment shows that the mechanical dynamic magnification, or Q, of a typical aerial array is about 60 times (although if it were welded together the Q might reach 100). This means that the amplitude of deflection produced by the continuous application of the vortex shedding forces will be 60 times as great as would be produced by the single application of a static force of the same size. The stresses in the aerial are likewise 60 times as great for the vibrating case as they are when the same forces are applied but there is no vibration.

The natural frequency will be exactly matched by the vortex shedding frequency at the 'awkward' wind speed and Q will be 60. If however the lowest natural frequency is different from the vortex shedding frequency being generated by the wind, the dynamic magnifier will be reduced to a value:-

$$Q = 1 / \sqrt{[1+r^4-r^2[2-1/60^2]]} \qquad (3)$$

where r is the ratio of forcing frequency to natural frequency.

It is possible to calculate the amplitude of any vibration of an aerial in the wind. The procedure is to find the band of vortex shedding frequencies for the mast and any of the elements using equation (1) and confirm either that there is a natural frequency within that band or to find the nearest corresponding natural frequency above that band. This amounts to finding the lowest natural frequency using the coefficients given in the appendix. If the natural frequency lies within the vortex shedding frequency band, Q will be 60. Otherwise, the ratio of the highest vortex shedding frequency to the natural frequency can be found and thus the value of Q from equation (3). The value of the force applied to the aerial can be determined from equation (2) and the elastic deflection for a static force of this size, again using the data given in the appendix. Multiplying the static deflection by the magnifier Q will give the required amplitude of vibration. The dynamic stresses can likewise be determined.

All this calculation is a lot of trouble but if a radio amateur were embarking on an expensive new multi-element array to be set up on a mountain somewhere, it might be worthwhile to do this calculation and design out any problems before a single element is cut. Most of us, however, merely wish to cure that noisy vibration on the roof, or try to make the array last longer than the first six weeks of every winter. In this case we can short cut the calculations and rely on Murphy's Law to ordain that there will surely be an 'awkward' wind speed which we cannot prevent by design and which will impair our aerial unless we can circumvent it.

Methods of reducing problems.

Wind excited vibration can be reduced or even prevented by:-

1. Making members less liable to easy vibration
2. Using spoilers on any members at risk to prevent vortex shedding from the whole member at the same time
3. Incorporating more damping

and in every case by using common sense like at least pointing a Yagi in the best direction during a storm.

IMPROVING THE STRUCTURE

Cantilevers with one end free are notoriously easy to vibrate and a Yagi array is just a collection of cantilevers fastened to a boom. There are seldom problems at vhf where the cantilevers are short and stiff but the longer cantilevers of lower frequencies are vulnerable. An element which visibly sags can be stiffened by supporting the ends with a cord to a support point above the boom, or by passing the cord through the element and using the tension in the external horizontal cord to support the ends as in **Figs 2 and 3.** Both are preferable to drooping elements if there is any sign of vibration. Often an array can be improved considerably by using thin cords as triangulated bracing between the various elements of the array. It is cheap, light and 'invisible' and it improves both aerodynamic and structural damping.

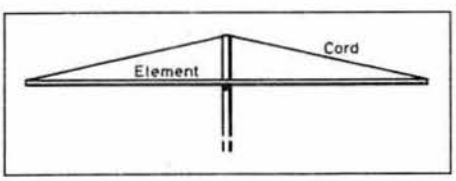


Fig 2.

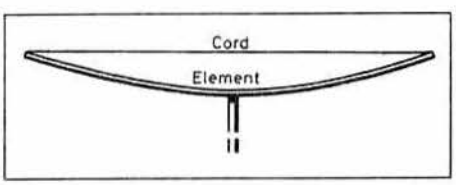


Fig 3.

Sometimes one particular mode of vibration is evident, for example, although the strength of an unguyed mast is quite sufficient, it does vibrate appreciably at some steady wind speed. Common sense tells us to guy the mast with three stays too prevent that mode. Very little strength will be needed in the guys if the mast really is sufficiently strong, and almost invisible guys might get rid of the trouble.

SPOILERS

Spoilers are of several kinds but the simplest and cheapest is to wrap a spiral of thickish cord from end to end of the member as in **Fig 4.** This prevents the simultaneous shedding of a vortex along its whole length. Wire is not as satisfactory as cord because of its weight and the likelihood of electrolytic corrosion. The 'strakes' often seen spiralling round the top third of steel chimneys are a luxury version of course. They are often a 'three start helix' with the pitch carefully chosen, but a vibrating oil rig under construction was stilled by wrapping a handy rope spirally round the vertical members as a stop gap until the tops were properly welded.

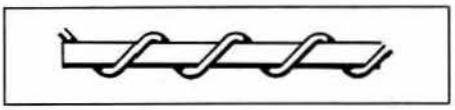


Fig 4.

It is possible to use a shroud fitted close to the end of an element and consisting of a hollow cylinder, about 10% larger in diameter than the element and about 10% of its length, punched with square holes (not round) and fitted over the element close to the ends. The air entering from the windward side, flowing round the element and out on the lee side, stabilises the element. They are troublesome to make in small diameters but large examples are sometimes seen near the top of steel chimneys.

Vortex shedding does not take place in a synchronous way unless the member has a clear uniform section at least 20 diameters long. Moreover, members of the 'wrong' diameter at any given wind speed, damp out much the same amount of energy aerodynamically as that which those of the 'right' diameter receive from the wind. It is possible to balance the positive excitation and the damping by a suitable mixture of element diameters as in **Fig 5.** The two diameters of the element must differ by about 25% and approximately equal frontal areas of the two diameters should be used.

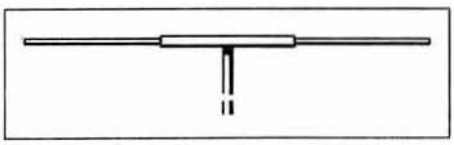


Fig 5.

The vortex street can be disrupted by using a fin on the downstream side of the member. Sometimes short tabs are used at intervals along the length and sometimes a single fin near the free end, **Fig 6.** The problem in this case is to make sure the aerial points in the right direction to make the fin 'downstream' when the awkward wind is blowing. Of course if the storms always blow up from one direction a fin might be a useful adjunct to a mast.

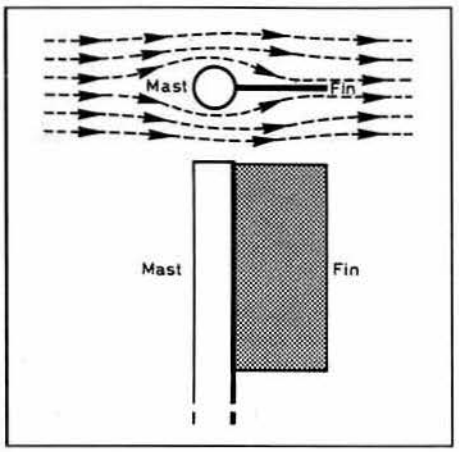


Fig 6.

DAMPING

Damping is a marvellous cure for vibration. The problem is to obtain dampers light enough and of the right range of strength for aerial arrays, and to find a suitable anchor for them. A car shock absorber is just a damper, of course, consisting of two eyebolts held together by a spring in parallel with a dashpot. The eyebolts apply external force from the part to be damped but the spring cannot be compressed or return to its original position without oil in the dashpot being forced through a small hole or round the edges of a slightly undersized piston. There is a fully worked out theory about how much damping to use but lorry shock absorbers rated for 10 tons have been used as dampers in structures. Few amateurs can have a stayed mast tall enough to merit this treatment but, if a mast were suffering vibration from vortex shedding, a suitable damper connected between the end of the stay and the ground could work wonders, **Fig 7.** Even a layer of soft fabric or other material between an aerial element and its clamp can make a difference.

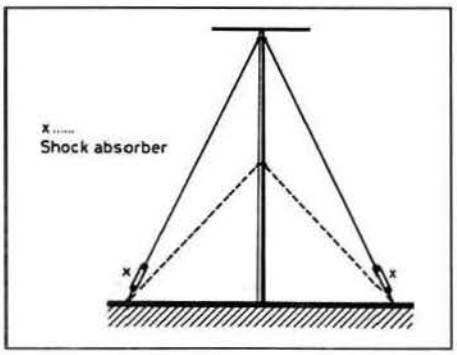


Fig 7.

Aerodynamic damping is equally acceptable and often easier to arrange. If a vibrating member carries members of other diameters across the airstream much damping can be gained. Imagine details of the 'flyswat aerial' appearing in which the owner swears that by soldering a wire mesh flyswat to each end of his Yagi elements he has not only cured all vibration but improved his SWR, the bandwidth of his array, and worked all the squares in Europe. Perhaps some of these advantages might be gained by ending each of the elements with a skeleton wire cone as in **Fig 8.** I vaguely recall a paper many years ago advocating this for electrical reasons,

although the length of the element has to be adjusted slightly. The aerodynamic damping of the elements is what normally prevents the boom of a Yagi vibrating when the wind is blowing at right angles to it.

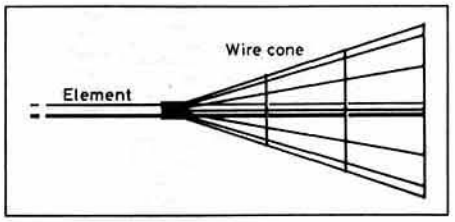


Fig 8.

G3VA in his *'Technical Topics'* (which actually inspired this article) has more than once drawn attention to another form of damper in which energy is dissipated by a piece of split hose or the like fixed near the end of elements [2].

Common sense is important, if there is a problem about vibration it is important to do something about it quickly. The calculations necessary for a mast carrying a boom supporting an array are given in the appendix and may help. All radio amateurs have learned some A.C. theory and mechanical vibration is merely a branch of audio oscillation, which can be tackled with confidence if we set our minds to it.

APPENDIX

1. Useful Data

Young's modulus, E,

Steel:	2.05×10^{11} N/m²
Aluminium alloy:	7.00×10^{10} N/m²
Copper:	1.03×10^{11} N/m²

The densities and ultimate tensile strength of the three metals are:-

Steel:	7830 kg/m³	430 N/mm²
Aluminium alloy:	2680 kg/m³	300 N/mm²
Copper:	8860 kg/m³	300 N/mm²

Maximum working stresses are about two thirds of the ultimate values.

2. Fatigue Stress

If the alternating stress can be kept below about 70 N/mm² for steel there will be no fatigue effects. Unfortunately there is no such fatigue limit for aluminium alloy or copper and fatigue damage will accumulate even at very low stresses. If the alternating stress can be limited to 20 N/mm², however, a satisfactory service life will be obtained.

The second moment of area, I,

For a thin-walled tube of outer diameter D metres and wall thickness to metres

$$I = \pi.t.D^3/8 \text{ m}^4$$

3. Natural frequencies

There is a wealth of information on natural frequencies, usually via differential equations, and matrix algebra. Most amateurs would prefer an approximation within about 10% of the 'exact' frequency if it could be obtained by simple arithmetic and this is what is shown below.

Basically, any vibrating system can be related to the case of a single mass vibrating on the end of a spring. In this case the natural frequency is:-

$$f_n = (1/2\pi) \times \sqrt{[\text{elastic stiffness/mass vibrated}]} \quad (4)$$

There are therefore only two things needed, the appropriate stiffness and the appropriate mass of the elements. The theory does not depend on the shape of the spring, it might be a leaf spring or a cantilever instead of a coiled spring; it could be a structure as in **Fig 9**, which you might otherwise have thought was one element of an aerial array supported by two booms. If

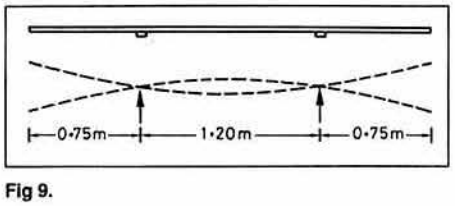


Fig 9.

Type of member	Frequency $\sqrt{(EI/pL^4)}$ X	Equivalent spring stiffness (EI/L^3) X	Equivalent mass (ME) (pL) X
Cantilever p kg/m, L metres	0·560	3	0·242
Cantilever end p kg/m, L metres		0	0·242
2 Pin ends p kg/m, L metres	1·571	48	0·493
Fixed/Pinned p kg/m, L metres	2·45	101·9	0·422

Table 1.

it vibrates in the mode shown it can be seen to consist of two cantilever ends and a central beam. We could find the stiffness of the centre of the member experimentally by hanging a weight of, say, 80 N there and measuring the deflection under the load. If this were 5mm (0.005 m) the stiffness would be the force applied divided by the deflection caused, ie 80 N/0.005 m or 16,000 N/m.

A table of stiffnesses to obviate the experimental effort for members often used is given in **Table 1.** The difference between cantilever ends and cantilevers in the table should be noticed. The cantilever end has no stiffness of its own, any stiffness it possesses is merely taken from the main member leaving the total unchanged. A true cantilever gains stiffness from its fixed support as does a double cantilever member, eg an element fixed only at its centre. The table allows you to work out the numerical values for tubes of different diameters, metals and wall thicknesses from the data given.

Now what about the appropriate mass? According to the theory the mass is vibrating on the end of the spring, ie where we measure the stiffness. This presents a problem because the mass of an element is distributed uniformly along its length rather than being concentrated at the point where we measure the stiffness. If the members were so light as to be of negligible self-mass and a 3kg mass were hung in the centre of the element, the frequency from equation 4 would be:-

$$f_n = (1/2\pi) \sqrt{(16{,}000/3)} = 11.6\text{Hz}$$

In fact the self-mass of the array is all important and separate masses are seldom applied. The appropriate mass mentioned earlier is that which, concentrated where we measure the stiffness, would have the same effect on the frequency as the distributed mass of the element. This can be calculated and is shown in Table 1. You will notice that for a cantilever the equivalent mass applied at the end is about one quarter of the self-mass and for a member supported at both ends about a half of the total self mass applied at the centre.

Occasionally there will also be a concentrated mass to consider as, for example, when a cantilever mast carries a rotary array on the top. In this case, add the concentrated mass of the array to the equivalent self-mass of the mast to obtain the total equivalent mass. Sometimes there will be more than one member, for example, when a mast is guyed at two levels. There are then in essence two vertical members, the lower with one end fixed in the ground and its top end pinned by the guys but free to rotate: and probably a cantilever end on top to carry the array. In this case add the separate stiffnesses and equivalent masses from Table 1 to obtain the total equivalents.

4. Stress imposed on the material

The maximum dynamic stress is obtained from F, the vortex force given in equation (2), the dynamic magnifier, Q, the length, L, of the member, its I value, and its diameter, D.

The formula for a cantilever, or a cantilever end is:-

$$\text{Stress} = [F \times L^2 \times Q \times D] / [4 \times I] \quad (5)$$

The formula for a member supported at both ends is:-

$$\text{Stress} = [F \times L^2 \times Q \times D] / [16 \times I] \quad (6)$$

If you substitute for F, Q = 60, and I in (5) you will find the stress is:-

$$= [40 \times (v \times L)^2] / [\pi \times t \times D] \text{ N/m}^2$$

$$= [12.73 (v.L)^2 / (D.t)] \times 10^{-6} \text{ N/mm}^2$$

The corresponding stress for a member supported at both ends is:-

$$= [3.18 (v.L)^2 / (D.t)] \times 10^{-6} \text{ N/mm}^2$$

Since the lengths of the cantilever end and the part between the two supports will be different, it is necessary to calculate both stresses and choose the higher. An element with two supports and two cantilever ends has its 'middle of the span' stress reduced by the amount of the cantilever end stress because of the continuity. The effective fatigue stress is then the larger of the cantilever end stress and this reduced central stress.

5. The amplitude of vibration

The amplitude of vibration of the end of a cantilever is:-

$$F \times Q \times L^4 / 8 \times E \times I \quad (7)$$

where all the terms have been defined before and their values can be substituted for particular members. The amplitude of the centre of a member simply supported at both ends is likewise:-

$$5 \times F \times Q \times L^4 / 384 \times E \times I \quad (8)$$

5. Examples

1) A steel mast of 65mm external diameter and 6mm wall thickness, 12m high and carrying an array of total weight 25kg.

a: Member properties

$$E \text{ value} = 2.05 \times 10^{11} \text{ Nm}^{-2}$$

$$\text{Mass per metre} = \pi \times 0.065 \times 0.006 \times 1 \text{ m} \times 7830 = 9.59 \text{ kg/m}$$

$$I \text{ value} = \pi \times 0.006 \times (0.065)^3 / 8 = 6.47 \times 10^{-7} \text{ m}^4$$

$$EI \text{ value} = 1.326 \times 10^5 \text{ Nm}^2$$

b: Maximum vortex shedding frequency

$$F_v = 45\text{m/s} \times 0.20 / 0.065\text{m} = 138\text{Hz}$$

c: Natural frequency

$$\text{Equivalent spring stiffness} = 3EI/L^3 = 3 \times 1.326 \times 10^5 / 12^3 = 230 \text{ N/m}$$

$$\text{Equivalent mass} = 0.24 \times p \times L + \text{concentrated mass} = 0.24 \times 9.59 \times 12 + 25\text{kg} = 52.9\text{kg}$$

$$\text{Lowest natural frequency} = (1/2\pi) \times \sqrt{[(230)/(52.9)]} = 0.332\text{Hz}$$

It is seen that the natural frequency lies within the possible band of excitation which stretches from zero to 138Hz. A check should therefore be made on any liability to fatigue and the amplitude of vibration. For stresses and amplitudes of vibration only the dynamic forces are concerned since these are what force the vibration, ie the steady wind drag force is not included. The dynamic stress produced by vortex shedding along the 12m mast is from equation (5):-

$$= [12.73 (v.L)^2/(D.t)] \times 10^{-6} \text{ N/mm}^2$$

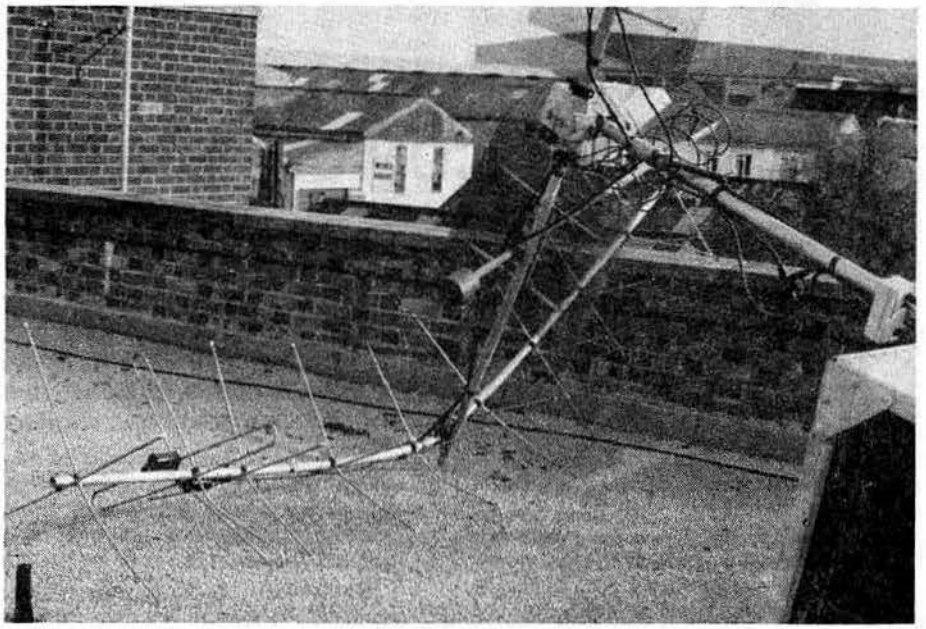
where v is the velocity driving the lowest natural frequency and can be obtained by simple proportion. If 138Hz is the vortex frequency produced by a wind speed of 45m/s then the operative 0.332Hz will be produced by a wind speed of 45 x 0.332 / 138 = 0.108m.s. The dynamic stress in the steel tube at its base is therefore:-

$$[12.73 \times (0.108 \times 12)^2/(0.065 \times 0.006)] \times 10^{-6} \text{ N/mm}^2 = 0.05 \text{ N/mm}^2$$

This is far too small for any fatigue damage but there is still the need to check the amplitude of vibration by equation (7). Substituting for F, Q = 60, L and EI gives:-

$$0.108 \times 60 \times 12^4 / 8 \times 1.326 \times 10^5 = 0.127 \text{ m} = 127\text{mm}$$

This is rather large for comfort although it is safe as far as the stress is concerned and commercial specifications often allow a deflection of 1% of the mast height. It would be wise to guy the mast and recalculate the frequency, stress and amplitude for the new configuration.



This article arrived just before the Jan 25 storm. As the photograph shows, we didn't read it in time!

2) The same mast but stayed at its top and mid-point with light guy ropes. It therefore consists of two members each 6m long with the lower end fixed in the ground. Using Table 1:-

$$\text{Total equivalent stiffness} = (48 + 101.9) \times EI / L^3 = 149.9 \times 1.326 \times 10^5 / 6^3 = 92022 \text{ N/m}$$

The self-mass of the array is now fixed by the top stay, therefore:-

$$\text{Total equivalent mass} = (0.493 + 0.422) \times p \times L = 0.915 \times 9.59 \times 6\text{kg} = 52.65\text{kg}$$

$$\text{Lowest natural frequency} = (1/2\pi) \times \sqrt{[(92022)/(52.65)]} = 6.66\text{Hz}$$

The dynamic stress produced by vortex shedding along the 6m span is from equation (6):-

$$[3.18 (v.L)^2 / (D.t)] \times 10^{-6} \text{ N/mm}^2 \text{ where } v = 45 \times 6.66/138 = 2.17\text{m/s}$$

The dynamic stress in the centre of the 6m upper span is therefore:-

$$[3.18 \times (2.17x6)^2 / (0.065x0.006)] \times 10^{-6} \text{ N/mm}^2 = 1.38 \text{ N/mm}^2$$

Again this is far too small for any fatigue damage.

The amplitude of vibration from equation (7) gives:-

$$5 \times F \times Q \times L^4 / 384 \times E \times I = 1.54 \times 10^{-4} \text{m or } 0.15\text{mm}$$ and quite satisfactory.

REFERENCES

[1] 'Wind Loading', DJ Reynolds, G3ZPF, *Radio Communication* April/May 1988, pp 252-255, 340-341.

[2] 'Technical Topics', Pat Hawker, G3VA, *Radio Communication* November 1989, pp 36-37.

Book Review

GUIDE TO BROADCASTING STATIONS

Edited by Philip Darrington

Published by Heinemann Newnes: ISBN number 0 434 90309 4

THE 1989/90 EDITION of this popular handbook has been on the bookstalls for several months now, but we have only just got around to reviewing it. It is edited by Philip Darrington, but has chapters written by several leading personalities and is well worth a place on your bookshelf, even if you are not keen on broadcast listening.

It successfully combines useful introductory material for anyone new to Short-Wave Listening, and provides a formidable list of long-wave, medium-wave and short-wave broadcast stations. It also provides a useful alphabetical list of which countries have transmitters on which frequencies, an example being that anyone looking for a transmission from the Cape Verde Islands is told that 3,931kHz is the place to monitor. Going to the other extreme, it is useful to know that if you are looking for Radio Moscow, you have 406 frequencies to choose from! There is also a worthwhile section which tells you when a particular country beams its transmissions to the United Kingdom.

Let us take a slightly more detailed look at the first 50 pages which act as the guide for the beginner. Firstly, it looks at the world of broadcasting outside the UK, offering the view that with so many powerful transmitters here, most people only find the occasional interference from an adjacent broadcast station an annoyance. However, those interested in broadcast band listening would be more likely to try to receive the 'interfering' station rather than the local, stronger one.

Just like broadcast stations here, foreign stations will transmit news and current affairs programmes, but will encourage you to listen longer by broadcasting feature programmes about their corner of the globe. There are also stations that feature special programmes for the SWL, and provide technical tips, equipment reviews and DX information.

For those interested in getting into BC listening, there is a useful table showing which receivers are on the market and their capabilities. From this it would be possible to form a view about which receiver you might wish to purchase. The details are only brief, but they give a hint of the capabilities of the sets.

Pat Hawker, who needs no introduction, provides a fine in-depth general look at listening, introducing us to the frequencies, conditions, interference, fading and distortion, aerials and propagation, and signal identification.

Most SWLs, whether amateur or broadcast enthusiasts, find most pleasure in sending reception reports and getting QSL cards. The next section is a must as it provides a very detailed insight into what BC SWLs should concentrate on when sending reports to stations. In broadcast circles, some of the QSLs (and pennants) on offer for a useful reception report are superb colour cards which are well worth displaying on the shack wall. It advises SWLs to make their reports stand out, but in the BC business it seems that some DX clubs offer standard printed forms - not at all appropriate for amateur reports, and, I imagine, not very easy to make them 'stand out'. However, the chapter has some useful advice which will obviously benefit the BC SWL.

The final section before the list of frequencies and stations is one which describes loop antennas. This will be of particular interest to those BC SWLs who cannot put up external antennas.

The guide has obvious advantages to the SWL who specialises in listening to broadcast stations, as it provides a comprehensive list of frequencies and the stations which use them. It is in its twentieth year and will quite obviously continue for many years yet.

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